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DESIGN STUDY OF SHAFT FACE SEAL WITH SELF-ACTING LIFT AUGMENTATION

III - Mechanical Components

by Lawrence P. Ludwig and Robert L. Johnson

Lewis Research Center
Cleveland, Ohio 44135

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16. Abstract

A main-shaft face seal with self-acting geometry was designed for gas turbine engine applications. The seal design goal was to minimize thermal deformation; this was achieved, in part, through use of thermal shielding and through use of a molybdenum alloy which has high thermal conductivity and low thermal expansion. Also, the seal seat was structurally isolated from the shaft by a radial spacer which mitigated the thermal movements of the shaft. Further, axial clamping of the seat through a bellows spacer limited the axial clamping force and thus minimized clamping deformation. Wear measurements after 120 hours of operation under simulated engine conditions of a 200-psi (138 N/cm^2) pressure differential, a 400-ft/sec (122-m/sec) sliding speed, and a 1000° F (811 K) sealed gas temperature showed that thermal deformation was not excessive.

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SUMMARY

A main-shaft face seal with self-acting geometry was designed for aircraft gas turbine engines. The self-acting geometry provides lift forces to separate the primary sealing surfaces at the operating speed. Therefore, seal wear, as a result of rubbing contact occurs only during startup and shutdown. The amount of lift is on the order of 0.0004 inch (0.0010 cm).

Since the seal design goal was to operate without rubbing contact, the seal force balance had to be controlled within the load limits of the self-acting geometry. Thermal deformation of the sealing faces is known to affect seal force balance; therefore, the design goal was to minimize these deformations. This design goal was reached, in part, through use of molybdenum alloy in the seat, retainer ring, and carrier. The molybdenum alloy has a thermal deformation that is about 1/7 that of conventional seat materials. Thermal deformation was further mitigated by reducing thermal gradients with thermal shields and with thermal isolation of the seat. In addition, the seat was structurally isolated from the shaft by means of a spacer that reduced the deformation effects of the shaft thermal growth. Also, by clamping the seat with a bellows, a predetermined axial clamping force was obtained, and deformation due to assembly clamping was minimized.

An acceptably low seal thermal deformation was demonstrated in 120 hours of operation at a 200 psi (138-N/cm^2) pressure differential, 400-foot-per-second (122-m/sec) sliding speed, and 1000° F (811 K) sealed gas temperature. Inspection of the surfaces after 120 hours of operation revealed that the primary sealing faces had an acceptable deformation (evidenced by the lack of tapered wear) as had been predicted by analysis. The calculated average operating sealing gap based on gas leakage of 12 standard cubic feet per minute (0.31 std m^3/min) was 0.0004 inch (0.0010 cm). Operation with rubbing contact at the very small film thickness is further evidence of an acceptable sealing face deformation.

INTRODUCTION

Aircraft gas turbine engines have main shaft seals to restrict gas leakage into the bearing compartments (sumps). Various types of shaft seals are used. Some engines use labyrinth seals exclusively, others use face or circumferential contact seals. Labyrinth seals have disadvantages of high leakage rates and associated debris passage; long operating life and reliability are the chief advantages compared with face and circumferential type seals which have low leakage capability. But contact seals are limited in pressure differential and speed because of rubbing contact. For conventional face contact seals, current limits are near a 125-psi (86.1-N/cm^2) sealed pressure differential, a 350-foot-per-second (107-m/sec) sliding speed, and a 800° F (700 K) sealed gas temperature (ref. 1).

Recent developments in seals with self-acting lift augmentation (ref. 2) promise a seal with the speed capability of labyrinth seals and the low leakage capability of contact seals. This self-acting lift seal is similar in construction to that of a conventional face seal except that the self-acting geometry acts to keep the sealing surfaces separated, thus the seal has high speed potential. Rubbing occurs only on startup and shutdown. The lift force of the self-acting geometry also provides positive gas film stiffness (ref. 3) which allows the seal head to dynamically track the runout motions of the seat face. Further, because of high gas film stiffness the self-acting geometry inherently operates with a very small gas film separation. As a result the mass leakage flow through the seal is much less than that of a labyrinth seal.

A vital design consideration in the self-acting face seal (or conventional contact seals) is the relative displacement of the sealing surfaces due to body forces, surface forces, and thermal growth. The relative displacement of sealing surfaces can have a significant effect on the seal force balance (ref. 2), which is vital in high-speed and high-pressure operation. The overall objective of the seal design therefore was to obtain a seal suitable for operation in advanced gas turbine engines by minimizing the relative displacement of the sealing surfaces. The more specific objective of this work were to (1) describe the mechanical seal design considerations that were made to minimize the relative displacements of the sealing surfaces, and (2) determine qualitatively that, under simulated engine operation, seal deformation is within an acceptable level.

The design analysis was applied to a full-scale mainshaft seal with a 6.60-inch (16.76-cm) mean sealing diameter. The design was for operational cruise at a 150-psi (103-N/cm^2) sealed pressure differential, 500-foot-per-second (153 m/sec) sliding speed, and 800° F (700 K) sealed gas temperature. In the design analysis, calculated thermal maps, stress fields, and displacement fields were used in the design refinements leading to the final selection of materials and geometry. This report is part III of a series of reports (refs. 3 and 4) on the self-acting seal that was built and run at simulated engine conditions under contract NAS3-7609. (See ref. 1 for rig description.) Reference 3 covers the design of the self-acting geometry, and reference 4 considers the design of the primary seal.

NOMENCLATURE FOR FACE SEALS WITH SELF-ACTING GEOMETRY

Within the seal industry there is a wide variety of terms used to describe similar seal parts. The ASLE seal glossary (ref. 5) has provided much needed guidance in seal nomenclature. This paper proposes a seal nomenclature that is descriptive of seals with self-acting geometry and also general enough so that it applies to face seals as a group. This self-acting nomenclature is mainly an extension of the work begun in reference 5. The suggested terms are shown in figure 1 which is a partial cross section of a face seal with self-acting lift geometry. As with a conventional face seal it consists of a rotating seal seat (all rotating parts are shaded) which is attached to the shaft and nonrotating seal head that is free to move in an axial direction thus accommodating axial differential movement between rotating and nonrotating seal parts. For example, differential thermal axial expansion of the engine in some cases causes axial movements (differential) of 0.25 inch (0.61 cm). In operation, the primary sealing faces are separated by a gas film thickness (in the range of 0.0004 in. (0.0010 cm)). This positive separation is a result of the balance of seal forces and of the film stiffness of the self-acting geometry. The self-acting geometry can be any of the various types used in gas thrust bearings; the Rayleigh step type were used for this study and are described in detail in reference 3.

The proposed face seal nomenclature is described in the next sections.

Nomenclature Applying to an Assembly of Parts

(1) Primary seal - Seal formed by the sealing faces of the seat and primary ring. Relative rotation occurs between these sealing faces.

(2) Secondary seal - Seal formed by the sealing faces of the secondary ring. In the case of a bellows seal the secondary seal is the bellows itself.

(3) Static seal - Seal formed by the mating surfaces of the primary ring and its carrier.

(4) Self-acting geometry (the term hydrodynamic geometry is used for incompressible fluids) - Lift pad geometry and mating face which together produce the thrust bearing action to separate the sealing surfaces.

(5) Film thickness (h) - Distance between primary sealing faces or between surfaces forming the self-acting geometry. (Note that h may vary with radial or circumferential position and with time.)

(6) Seal head - Assembly that is axially moveable and consisting of primary ring, its retainer, (if any) and its carrier. (Note that the retainer and the carrier are combined into one part in some designs.)

Nomenclature Applying to Single Parts

(1) Seat - Part having a primary sealing face and mechanically constrained with respect to axial motion.

(2) Primary ring - Ring having a primary sealing face and part of the seal head.

(3) Retainer, primary ring - Ring that retains primary ring.

(4) Secondary ring - Part having secondary sealing surfaces which mate to the secondary sealing surfaces of a carrier.

SEAL MECHANICAL DESIGN

Seal Assembly

The final design configuration of the face seal with self-acting geometry is shown in figure 2. Gas leakage is from the high pressure side (inside of carbon primary ring) through the primary seal into the bearing compartment (sump). This gas leakage pressurizes the sump and assures proper scavenging of the bearing lubricant. It should be noted that although the sealed gas temperature is high, tests have shown that a considerable gas temperature drop occurs in the leakage flow so that the leakage into the sump has not posed a fire hazard when the seal was operating properly. Operation to 40 standard cubic feet per minute (1.05 std m^3/min) and 1200° F (922 K) sealed air temperature has been obtained without a sump fire.

Seat Assembly

Conventional face seal seats have construction similar to that shown in figure 3. The practice is to axially clamp the seat by means of a bearing lock nut (not shown). Thus, the seat, in effect, becomes rigidly attached to the shaft. Experience has shown that to prevent seat face deformation, the end faces of the shoulders (spacers) clamping the seat must be optically flat; the seat, of course, should be optically flat on both faces. A common assembly mistake is to clamp the seat with a spacer whose end mating faces are not flat. This spacer face out-of-flatness can then be transferred into the seat face during assembly since the axial clamping can readily reach several tons of force. Further, since the seat bore contacts the shaft, nonuniform shaft thermal radial growth (or thermal growth of a clamping spacer) can cause coning deformation of the seat face as illustrated in figure 4.

Figure 5 shows the final seat design for the self-acting face seal. The seat is thermally and structurally isolated from the shaft by the following means:

(1) The thin wall spacer between the seat and shaft (fig. 5) serves as a radial spring that mitigates the effect of nonuniform shaft thermal growth due to the axial thermal gradient. The effect of thermal movement is further attenuated by piloting the seat over its centroid.

(2) The bellows clamping spacer applies a predetermined clamping force (approximately 2000 lbf (8896 N)). This clamping force has proved to be large enough for proper assembly of the seat, as evidenced by the fact that face total indicator runouts of 0.0008 inch (0.0020 cm) are commonly obtained. The 2000-pounds force (8896 N) is probably an order of magnitude less than clamping forces produced by the usual bearing lock nut assembly technique and therefore the deformation during assembly is minimized.

The seat thermal design is tailored to minimize the axial thermal gradient. Use of a molybdenum alloy, instead of the conventional SAE 8740 or heat resistant alloy, provides relatively low deformation. The deformation criterion is defined as the ratio of thermal expansion to thermal conductivity (ref. 6). As an example, if the molybdenum is given a relative rating of one, SAE 8740 would have a deformation rating of seven, and the heat resistant alloys an even higher rating. Thus, the molybdenum alloy provides a significant advantage in lower deformation as compared with the conventional seat materials.

The cooling oil flow path is also tailored to produce the minimum axial thermal gradient. The cooling oil passing under the thin wall spacer contacts the hotter seat face (seat primary face) first. This oil route causes less of a thermal gradient than allowing first contact of the oil with the cooler seat face (face nearest to the bearing). Centrifugal pumping action due to rotation and the 45 radial exit holes near the hot face (seat primary sealing face) assures even distribution of the cooling oil. Figure 6 is a typical calculated thermal map (ref. 7) for the design operating conditions.

Inspection of the temperature field reveals that the heat shields are effective in reducing the thermal gradient in the seat. As an example, the heat shield, which protects the seat face from the oil passing through the bearing, has a calculated temperature of 427^{o} to 492^{o} F (493 to 529 K) the adjacent seat face is 651^{o} to 658^{o} F (617 to 621 K). Without this shield the seat face would be subjected to oil cooling, and the seat gradient and corresponding deformation would be significantly greater.

Figure 7 gives the calculated seat assembly stresses due to combined centrifugal and thermal effects. In general, about one-half of the stress magnitude can be attributed to thermal effects, the rest arise from centrifugal effects. Thus, the thermal gradient, in addition to being a major factor in seat deformation, is also a very significant factor in stress levels.

Figure 8 shows the calculated seat assembly displacement (in radial and axial directions) due to combined centrifugal and thermal effects. As is the case of the stresses, the thermal effects are significant. The important point is that the seat primary sealing face remains relatively square with the shaft centerline. The net calculated coning deformation of the primary sealing face is 0.0006 inch (0.00152 cm) in the axial (z) direction; this magnitude of deformation (which can be accommodated by the self-acting geometry as discussed in ref. 3) is equivalent to a 0.68-milliradian taper across the seat primary sealing face.

As shown in figure 2 the seat primary sealing face is exposed to hot air; a hard chromium carbide coating on this face provides wear and oxidation resistance. The face finish was 4 to 8×10^{-6} inches (10 to 20×10^{-6} cm) root-mean-square with the face flat within two light bands. A spiral groove pattern (oil windback) in the face of the seat (see fig. 5) prevents oil from leaking into the sealing dam area during operation.

Seal Head Assembly

The seal head assembly (see figs. 2 and 9) consists of a primary (carbon-graphite) ring which is shrink fitted into a molybdenum retainer ring. This assembly in turn is piloted (located) by a resilient ring attached to the primary ring carrier. The resilient ring and the static seal are important features in the prevention of thermal deformation. The resilient ring is flexible enough to allow differential thermal growth between the carrier and the primary ring. Thus the surfaces forming the static seal move (radially) relative to each other, and coning deformation is not induced into the primary ring. Molybdenum alloy is used in the carrier and retainer ring in order to minimize thermal deformation. The relatively high density of molybdenum, however, is a disadvantage in that the inertial forces associated with seal head movement are higher.

The carbon-graphite primary ring contains the self-acting geometry (lift pads) that are shown in figure 9 and described in reference 3. The carbon-graphite material is small grained and has a high amorphous carbon content (as compared with the percent of graphite) in order to obtain maximum strength and wear resistance. Measurements by thermocouples have shown that, when the sealed gas temperature is 1000° F (811 K), the carbon temperature is only 600° F (589 K) near the sealing surface. This significant difference from the sealed gas temperature is achieved in part by redirecting the cooling oil flow (leaving the seat from the 45 oil exit holes) to the head assembly outside diameter. (The oil baffle to redirect flow is not shown in fig. 2.)

Figure 10 is a thermal map of the head assembly for the design conditions (mean seat sliding velocity, 500 ft/sec (153 m/sec); sealed gas temperature, 800° F (700 K); oil temperature in the sump, 370° F (460 K)). These thermal gradients produce relatively low stresses in the assembly (see fig. 11).

The net closing force (due to the sealed pressure) on the primary ring is determined in part by the relation of sealing dam diameters to the secondary sealing diameter (see fig. 1). (See ref. 2 for a discussion of seal closing force.) Significant changes in closing force, therefore, can be associated with differential thermal growth of these diameters. An additional consideration is the force moment induced by eccentricity between the secondary sealing diameter and the sealing dam diameters. This eccentricity was minimized by means of the mechanically resilient piloting ring previously discussed (see fig. 2). This piloting ring contacts the carbon ring in the centroidal plane of the primary ring cross section. Measurements after seal assembly revealed that the eccentricity was less than 0.002 inch (0.005 cm).

The principal effect of the primary seal ring deformation is the formation of a divergent leakage gap at the sealing dam, as indicated in figure 12 which shows the calculated divergence of the primary seal ring. The calculated magnitude of the divergence is 0.5 milliradian across the self-acting lift pads. Thus the total divergence (primary seal ring and seat) is 1.18 milliradians. This is within the practical limits as discussed and analyzed in references 2 and 3.

The primary ring carrier (fig. 2) is made from molybdenum alloy for the purpose of matching the thermal growth of the primary (carbon graphite) ring and for the purpose of minimizing thermal gradients that would induce detrimental deformation. The secondary seal (fig. 1) is pressure balanced in both the axial and radial directions. It is also made from a molybdenum alloy and is chrome plated (as is the mating secondary sealing diameter) for oxidation and wear resistance.

OPERATING DATA

Thermal Deformation

The seal was operated under test rig conditions (ref. 1) that simulated an engine sump for 120 hours at a sealed pressure differential of 200 psia (138 N/cm^2), a sealed gas temperature of 1000° F (811 K), and a sliding velocity of 400 feet per second (122 m/sec). The cooling oil flow through the seal seat was 15 pounds per minute (6.8 kg/sec) and the oil-in temperature was 285° F (413 K). Thus the overall axial gradient across the seat was 650° F per inch (243 K/cm). Surface profile traces, taken after the 120 hours (fig. 13), indicate wear of less than 0.0001 inch (0.00025 cm) and no taper wear (taper wear would be evidence of a sealing face deformation). The shape of the surface profile trace (fig. 13) shows slight erosion rather than wear due to sealing face deformation. The land surface at the inside diameter of the self-acting lift pad had negligible wear; the land at the outside diameter of the pad had about 0.000100 inch (0.000254 cm) wear tapered to suggest convergent (in the direction of gas leakage) face deformation. However, the surface profile from the inside diameter to the middle of the sealing dam (primary seal face) suggests divergent faces. Thus, erosion, not wear from sealing face deformation, is postulated.

Gas Leakage

Figure 14 shows that the gas leakage during the 120 hours was near 12 standard cubic feet per minute (0.31 std n^3/min). Using the leakage as a basis, the calculated average operating film thickness is 0.0004 inch (0.0010 cm) and this is near that predicted in the design analysis (refs. 3 and 4).

SUMMARY OF RESULTS

A face seal with self-acting geometry was designed for operation in gas turbine engines. Design operating conditions were 150 psia (103 N/cm^2) sealed gas pressure, 500 feet per second (153 m/sec) sliding speed, and 800° F (700 K) sealed gas temperature. In a 120-hour test at 200-psi (138-N/cm^2), 400 feet per second (122 m/sec), and 1000° F (811 K), the wear was 0.0001 inch (0.00025 cm) with no taper wear. Further, the operating film thickness (based on leakage) was 0.0004 inch (0.0010 cm). The lack of taper wear and successful operation at very small film thicknesses reveals that the thermal deformation of the primary faces must be within acceptable limits as predicted by analysis.

Extensive use was made of molybdenum alloy in the design in order to minimize thermal gradients that cause detrimental seal deformation; in particular the seat, carbon retainer ring, carbon carrier, and secondary seal were made from the molybdenum alloy.

The seat was structurally isolated from the shaft by a radial spacer in order to mitigate the deformation effects caused by shaft thermal displacement. Further, the seat was clamped through a bellows which provided a predetermined amount of axial clamping and mitigated clamping distortions. Also seat axial thermal gradients which induce undesirable deformations were minimized by thermal shielding and oil cooling.

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio, November 12, 1970,
126-15.

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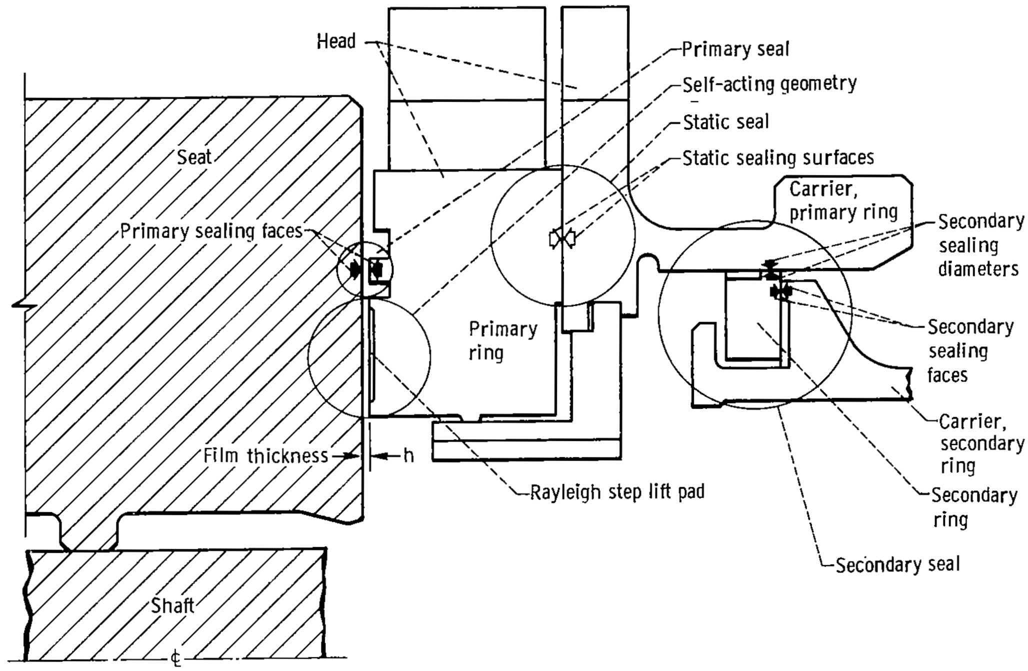


Figure 1. - Seal nomenclature for self-acting face seal.

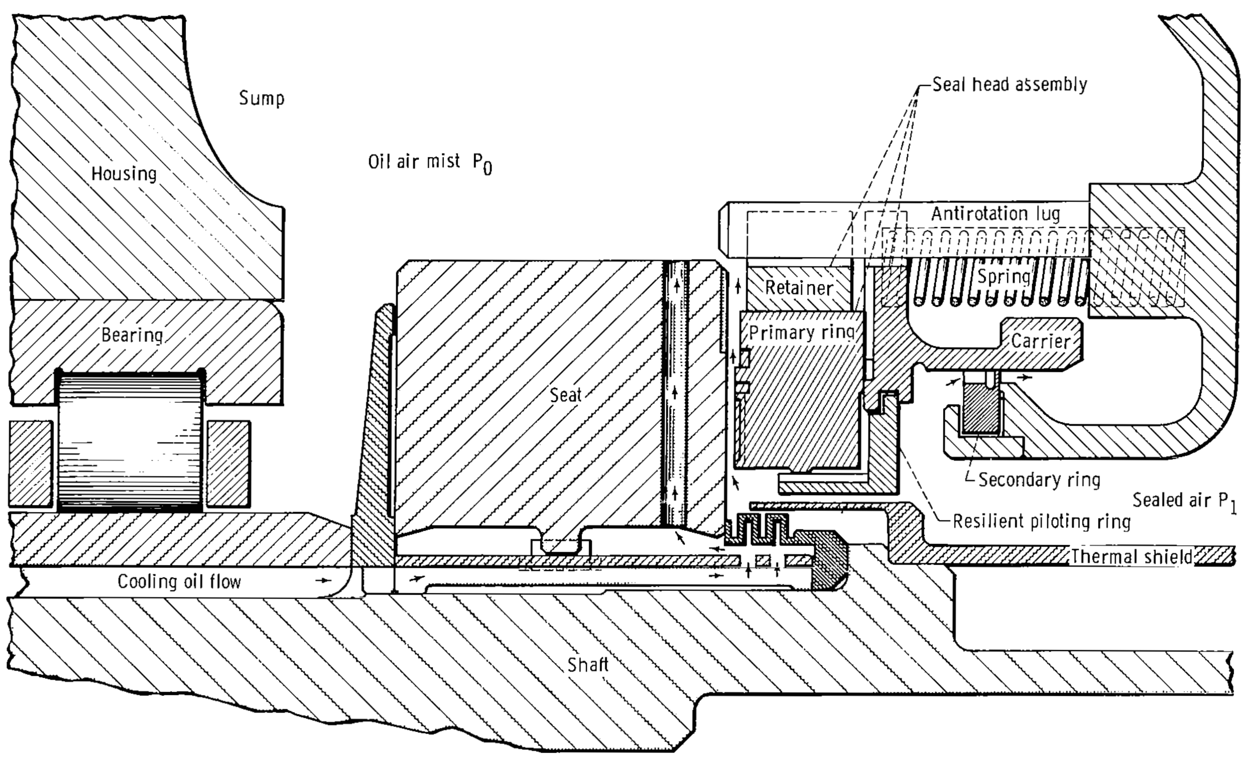


Figure 2. - Shaft face seal with self-acting lift geometry; sealed gas, 215 psia (148 N/cm^2 ab$_2$) (P_1), 1000° F (811 K); sump oil air mist, 15 psia (10.3 N/cm^2 ab$_2$) P_0, 285° F (414 K). (Note: rotating parts are shaded.)

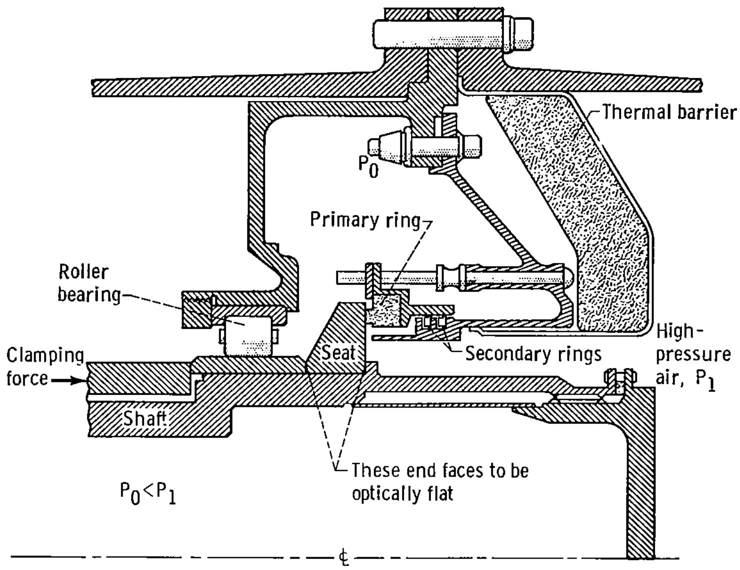
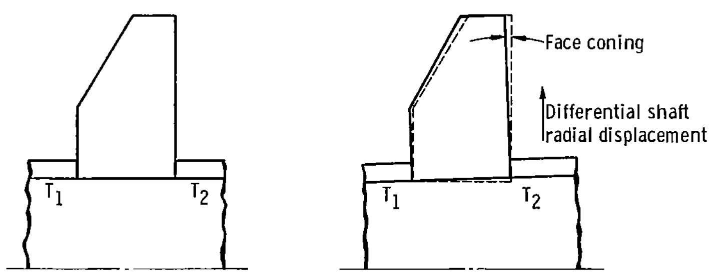


Figure 3. - Bearing and face contact seal assembly with conventional seat mounting practice.



(a) Isothermal shaft, $T_1=T_2$.

(b) Shaft with axial thermal gradient, $T_1<T_2$.

Figure 4. - Coning deformation of seat due to thermal displacement of shaft.

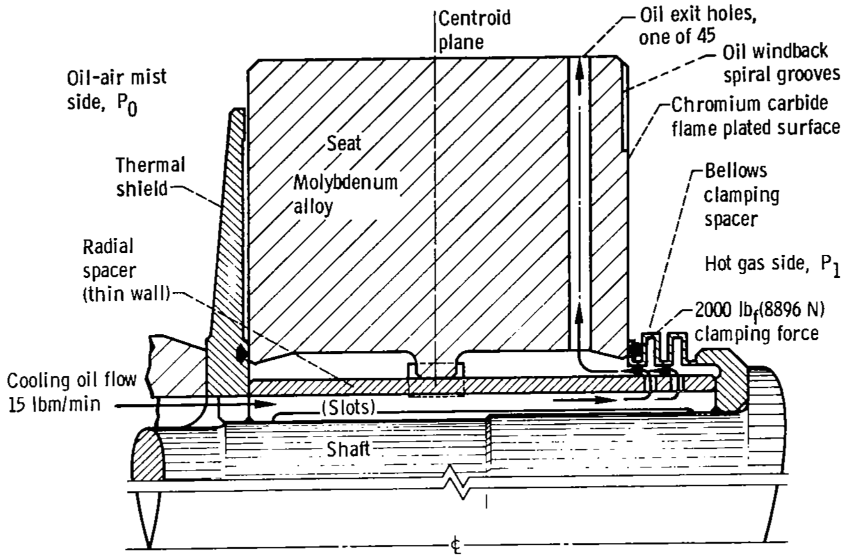


Figure 5. - Seat assembly for self-acting seal.

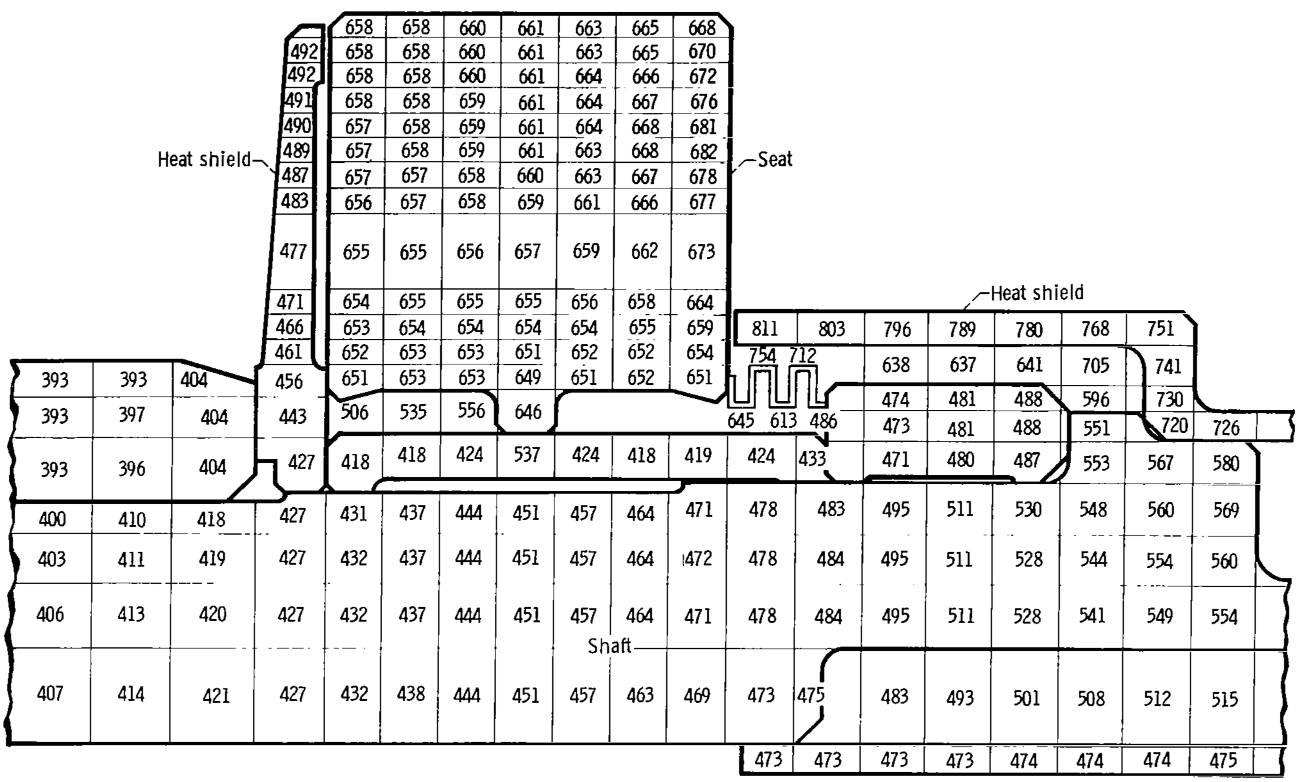


Figure 6. - Calculated temperature field (expressed in °F) of seat assembly. Design conditions: sealed pressure, 165 psia (114 N/cm^2); mean sliding speed, 500 feet per second (153 m/sec); sealed gas temperature, 800° F (700 K).

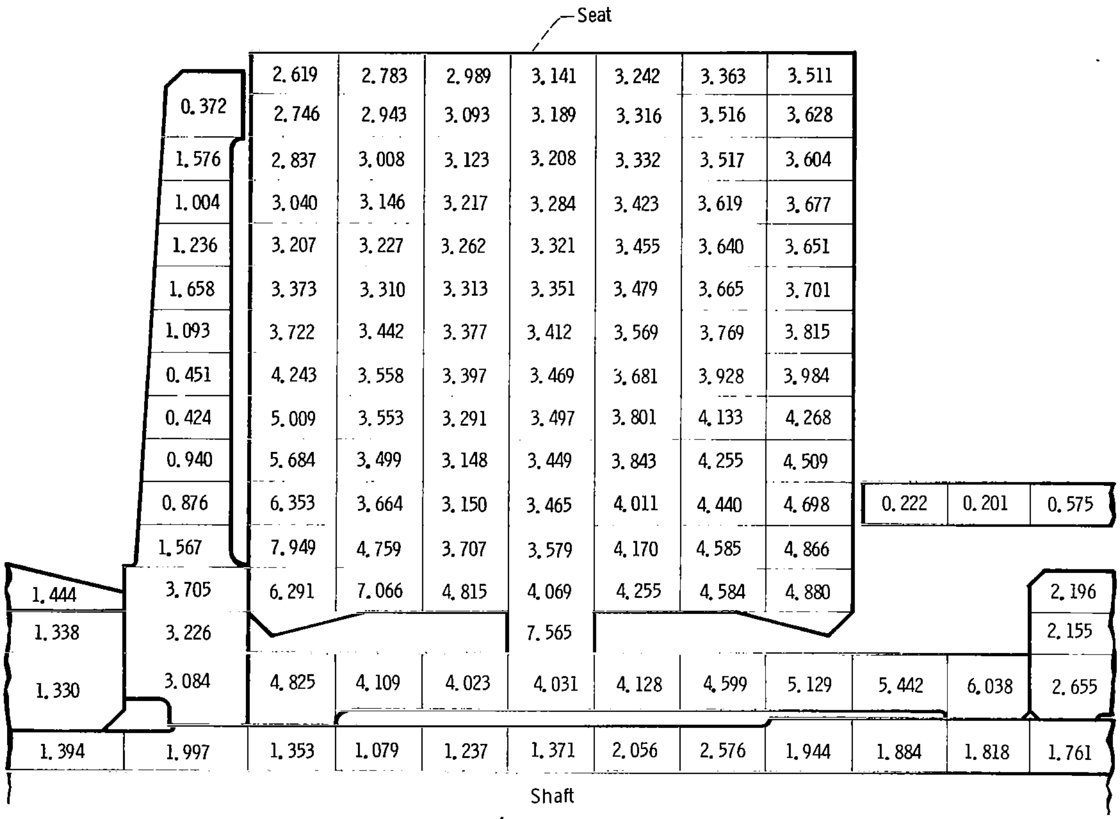


Figure 7. - Calculated stresses (expressed in psi x10^{-4}) in seat assembly due to combined centrifugal and thermal effects. Design conditions: sealed pressure, 165 psia (114 N/cm^2); mean sliding speed, 500 feet per second (153 m/sec); sealed gas temperature, 800° F (700 K).

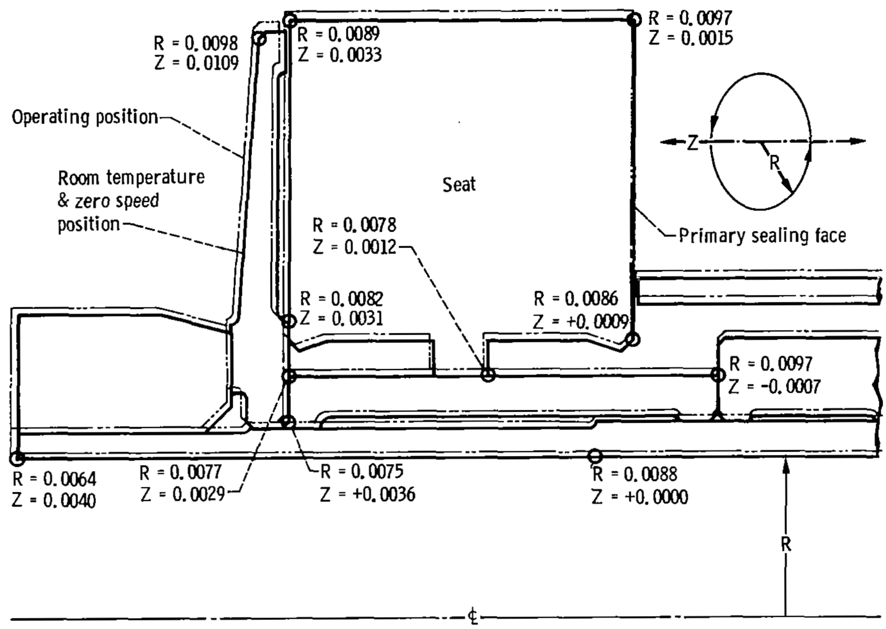


Figure 8. - Displacement of seat assembly due to centrifugal affects (displacements are expressed in inches). Design conditions: sealed pressure, 165 psi (114 N/cm^2); mean sliding speed, 500 feet per second (153 m/sec); sealed gas temperature, 800° F (700 K). (R denotes radial displacement; z denotes axial displacement.)

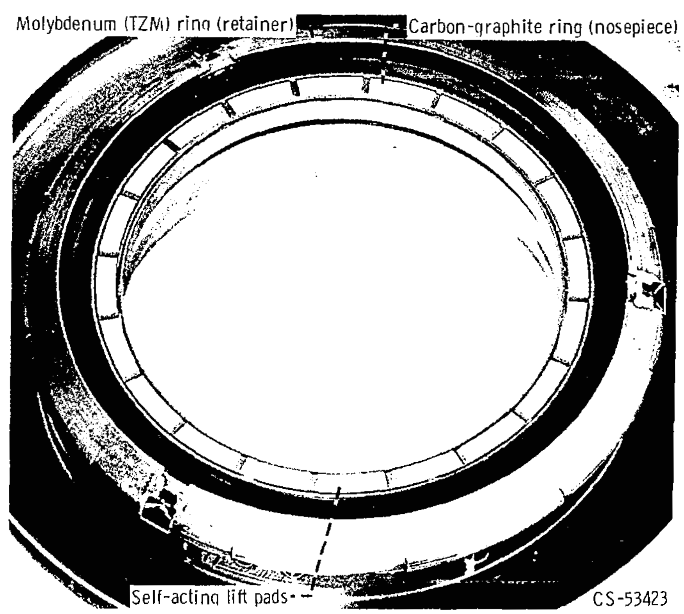


Figure 9. - Carbon-graphite and retainer ring assembly after 120 hours endurance running at 200 psi (138 N/cm^2) pressure differential, 400 ft/sec (122 m/sec) sliding speed, and 1000° F (811 K) sealed gas temperature.

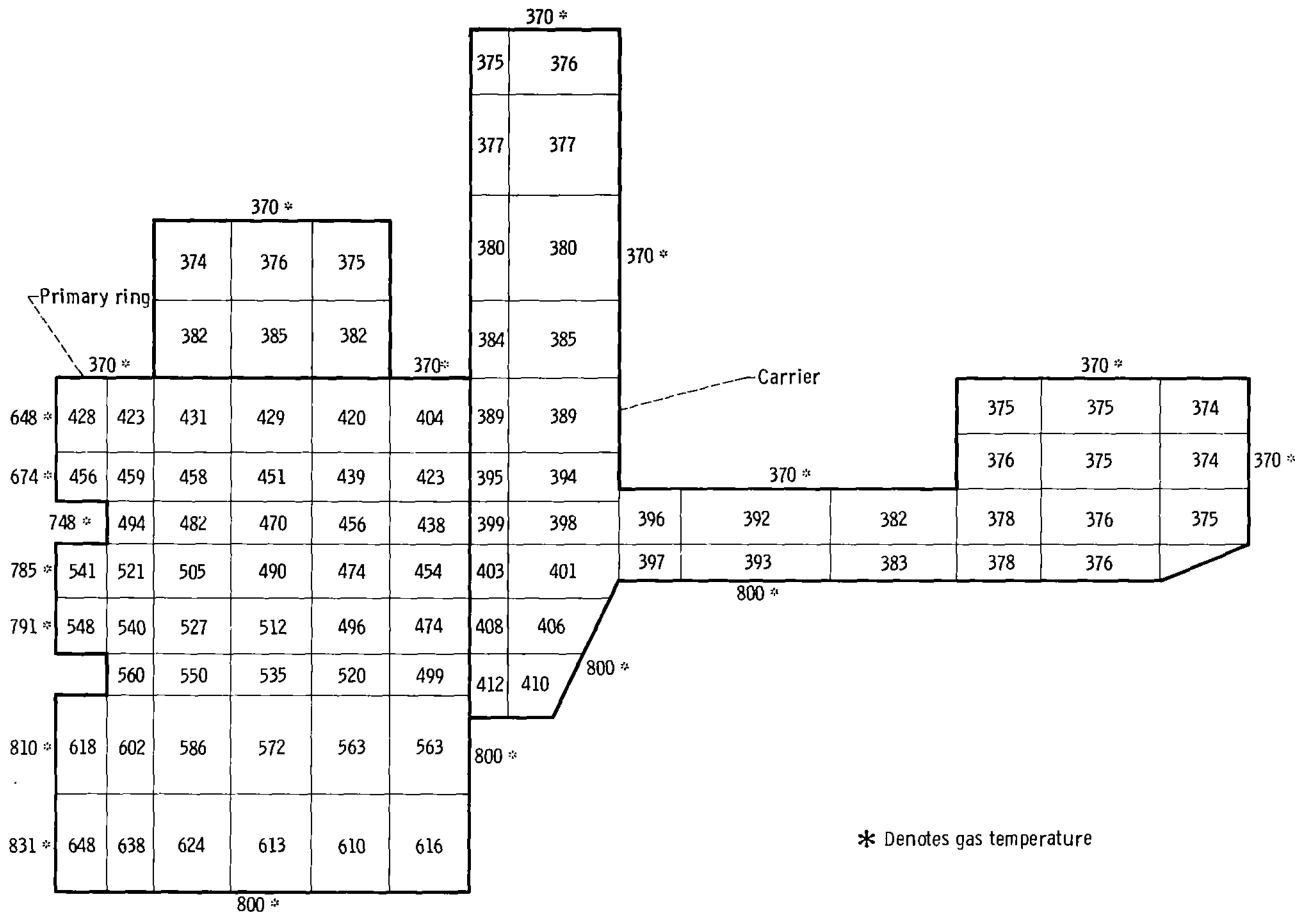


Figure 10. - Calculated temperature field of primary seal ring and carrier. (Temperatures are expressed in °F.) Design conditions: mean sliding speed, 500 feet per second (153 m/sec); sealed gas temperature, 800° F (700 K); oil sump temperature, 370° F (361 K).

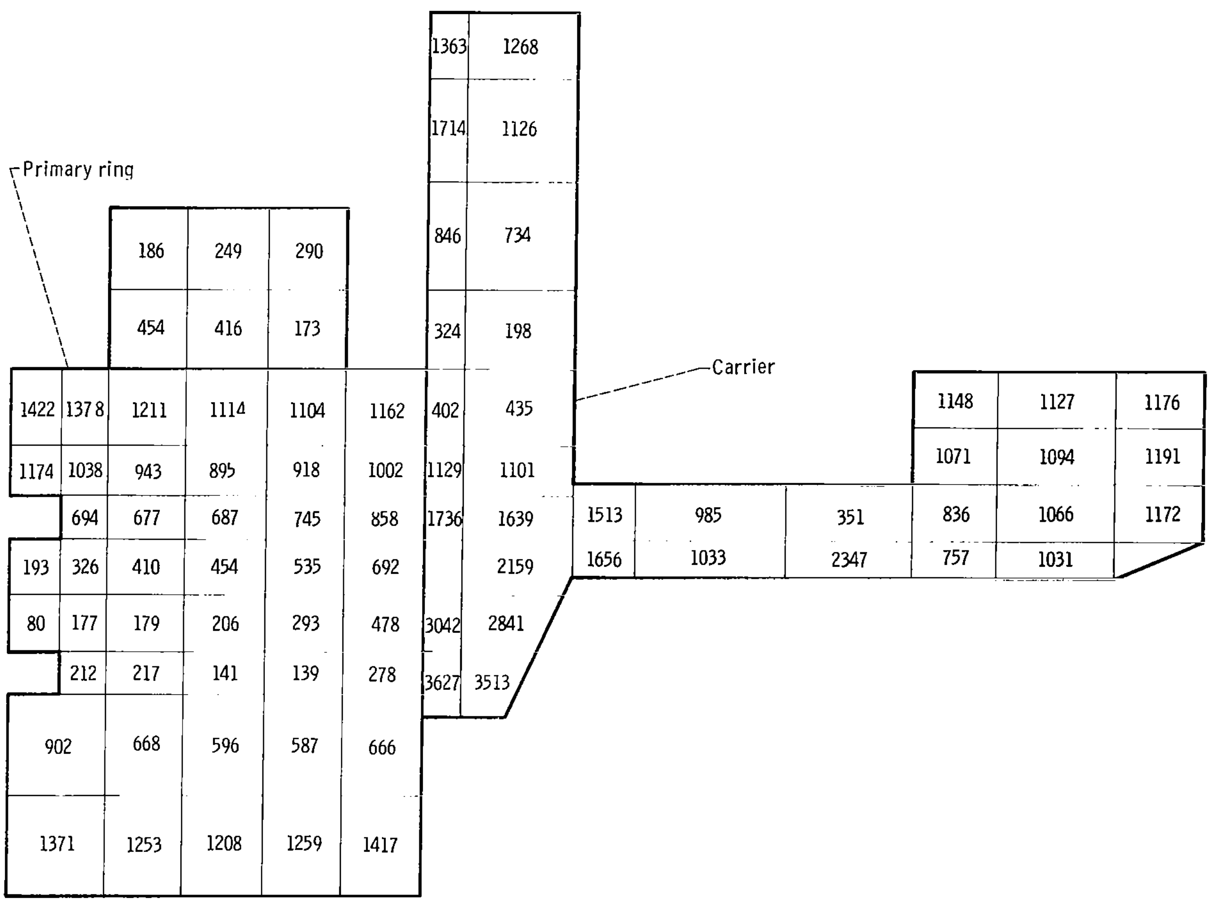


Figure 11. - Calculated stresses in primary ring and carrier due to temperature gradients. Design conditions: sealed gas temperature, 800° F (700 K); oil sump temperature, 370° F (461 K).

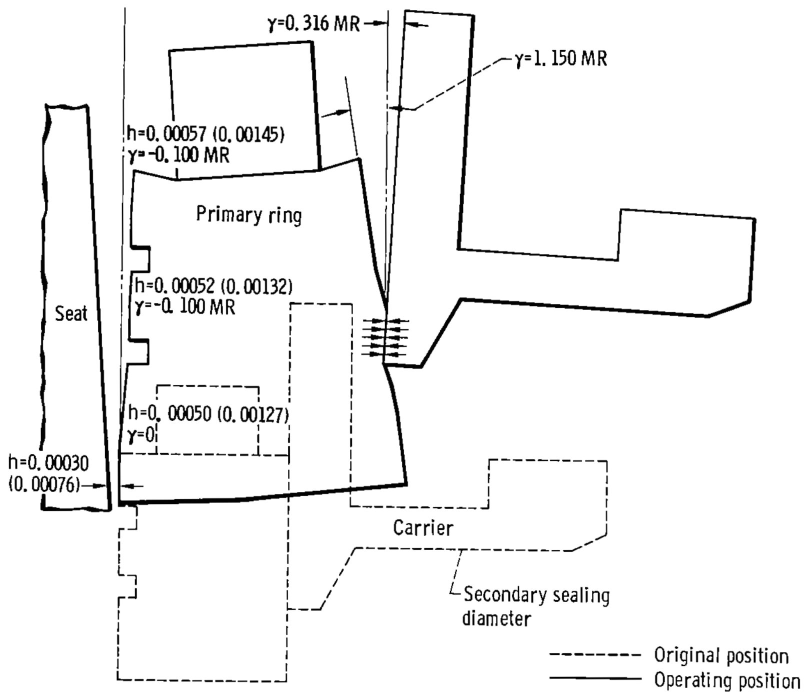


Figure 12. - Calculated displacement of primary ring and carrier assembly. Design conditions: sealed pressure, 165 psia (114 N/cm^2); sealed gas temperature, 800° F (700 K). Film thickness, h, is expressed in inches (cm); coning deformation, γ, is expressed in milliradians.

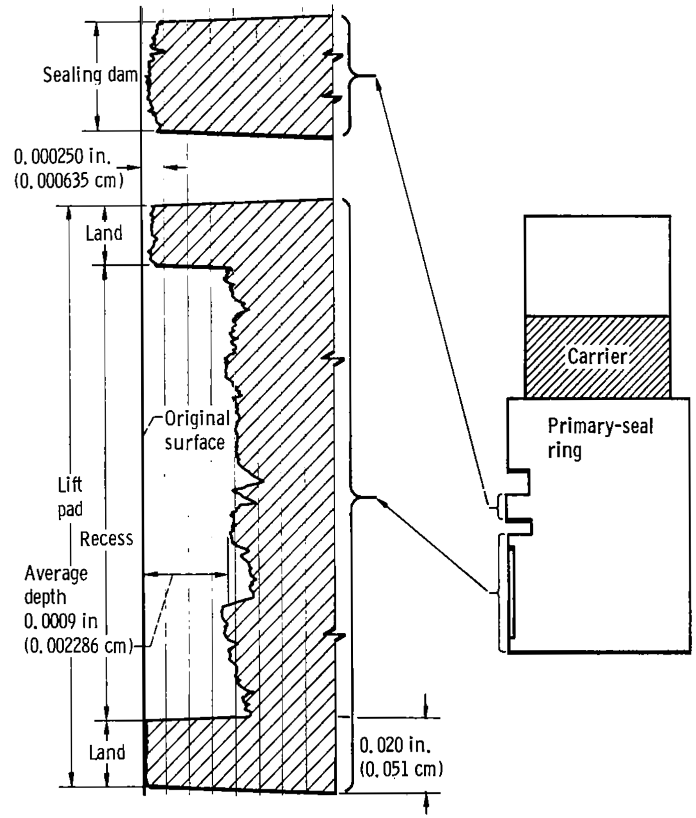


Figure 13. - Surface profile trace radially across lift pad and sealing dam after 120-hour run. Run conditions: sliding speed, 400 feet per second (122 m/sec); sealed pressure differential, 200 psia (138 N/cm^2); sealed gas temperature, 1000° F (811 K).

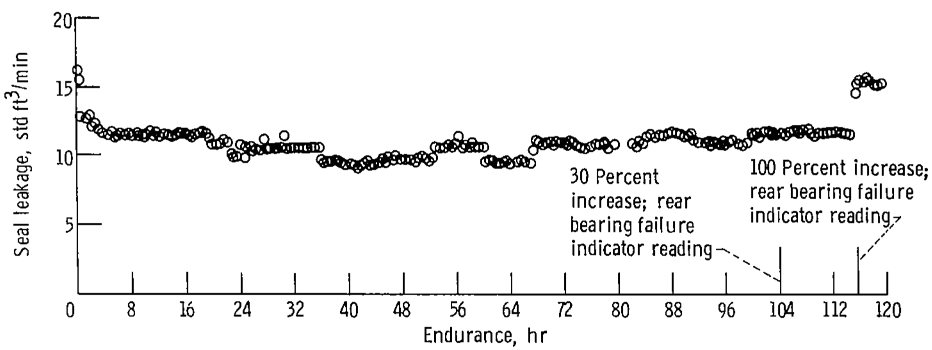


Figure 14. - Air leakage for 120-hour endurance run; sliding speed, 400 feet per second (122 m/sec); sealed pressure, 200 psi (138 N/cm^2); sealed temperature, 1000° F (811 K). Leakage increase after 115 hours was caused by vibration (bearing failure).

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